

NOISE ABATEMENT IN WOOD WORKING

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NOISE ABATEMENT IN WOOD WORKING

by

Wolfgang Schmutzler

Introduction - Noise sources - Measures to avoid noise:
Machines with cutting spindles; Circular saws; Suction removal
of chips; Cutting spindle mills - Evaluation of low-noise
intervals - Summary - Bibliography.

ABSTRACT: Manufacturers and users of machines are held by regulation and law to consider certain definite maximum values for noise in factories and workshops. On machines with cutting spindles, the sound intensity rises with the fourth power of spindle speed. Addition of ten equally noisy sound sources increases the total sound level by 10 dB. The main object in noise abatement is elimination of the noise source by design changes at the tool and the surrounding table and pressure lips. The propagation of noises is inhibited by attenuation of sound traveling through air and solids as well as by sound damping.

INTRODUCTION

Fast-running wood-working machines have long been unpleasant noise sources in working areas. The general development in the area of wood milling, increasing the spindle rotational rates and combining several processing units into one machine, brought a steady increase in noise production. In spite of this, noise abatement in wood-working machines has only rarely had an established place in the problem range of the designer. In most wood-working plants, therefore, the operating personnel are exposed to the danger of hearing damage. Circular saws and shapers are among the most unpleasant noise sources, especially if several processing units are combined into a single automated machine. In the chipboard industry the noise production of the mills is so high that the operators are exposed to great danger of hearing damage.

*Numbers in the margin indicate pagination in the original foreign text.

The long-standing requirement for the construction of low-noise machines was for a long time scarcely considered. Today, laws and regulations require the manufacturers and users of machines, devices, and equipment to consider definite maximum values for noise in and near working areas.

Noise abatement in wood-working machines is a difficult problem. Scientific bases and practical experience are lacking. Therefore, we attempt to give a review of the state of knowledge and to show some new potential solutions for decreasing noise. The difficulty of the problem will not allow us to arrive at appropriate solutions very quickly. Only through systematic research work will we be able to arrive at reliable information. The time until then must be bridged with solutions based on practical experience.

This work is primarily concerned with the noise produced by fast-running wood-working tools and with design measures for noise abatement. Kurtze (1964), Slawin (1960) and Zeller (1950) have reported on basic technological possibilities for noise abatement. Great advances have already been made in the study of the causes of noise in certain areas, which are also of interest for wood-working machines, as with electrical machines, engines, and machine frames (Heiss, 1950; Herrman, 1963, 1964; Jordan, 1950).

NOISE SOURCES

Ignoring the noise produced by secondary sources such as motors and drives, we can, with some simplification, reduce the working noise of running wood-working machines to three basic types of noise sources.

1. The production of sonic vibrations by the rotating tool. Causes which have been recognized for this include a pressure field rotating with the tool, periodic stagnations of air ahead of the tool cuts, axial tool motions, and inherent oscillations.

2. The tool striking the work piece. Here at least one of the two pieces is made to vibrate, giving off sound.

3. Production of sound vibrations by suction removal of chips. Aerodynamic noise arises from the flow process. Impacting chips cause vibrations in the collection hoods and pipelines.

Under operating conditions, these sources of noise always
occur together.

NOISE ABATEMENT MEASURES

Basically, we must start with abatement of the loudest noise source. Next, we should take care that the unavoidable noise is not propagated. We can differentiate the following principal measures:

1. Limiting the noise production by change of the working method (e. g., end mills instead of peripheral mills in working the thickness of chipboards) and by design changes in the tool, such as the table and pressure lips around the tool.

2. Barring sound conduction through solids, e. g., by interrupting the bridges conducting sound or by building in damping elements.

3. Damping airborne sound, as by additional sound-barrier construction elements, sealing machine openings by sound-blocking walls, etc., or by sound-damping design methods such as sound-absorbing locks at the inlet and outlet openings of the machine.

MACHINES WITH CUTTING SPINDLES

For successful noise abatement it is important to know the manner in which the noise generator works within the machine. Only in a few cases will noise abatement design be successful in eliminating the place where noise is produced. But when the designer knows the flow of noise within the machine, he has an important means to decrease the major radiation sites extensively. The pattern of the noise generator flow, e. g., in a thickness

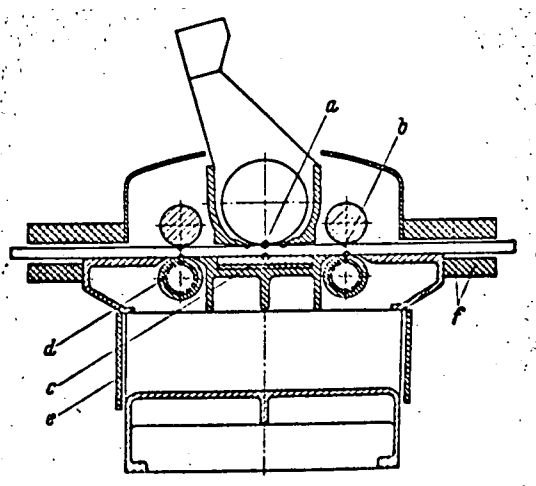


Figure 1. Noise sources in a thickness planer with noise abatement measures. a: primary noise source. b: secondary noise source. c: rubber plate in the table. d: rubber cushion on the feed roll; e: cover for the table opening; f: sound-absorbing lock.

planer, should show critical noise propagation points and serve as a basis for considerations on noise reduction (Figure 1).

The cutting spindle is the principal noise source. The noise propagation is limited by the cast clamping bars and by the feed roll covers. But the chip hood, which has a large surface and is usually made of thin sheet, also vibrates. Here, the vibration resistance is increased by material stiffenings, e. g., corrugations, and by damping high-frequency vibrations by means of an anti-drone mass. Steel sheet is not a practical material for chip hoods, from the viewpoint of noise abatement. Magnesium-aluminum alloys such as GMgAl6Zn DIN 1729 are better suited for this. They have good vibration resistance. But the effect of these measures must not be overestimated. In order that the solid sound conduction not propagate from the cutting spindle through the bearings to the machine stand, there should be insulation from an intermediate plate of pressure-resistant rubber.

Small improvements are also produced by noise-blocking covering of machine openings and noise-absorbing locks at the inlet and outlet openings. But success with these measures is highly dependent on the personal experience of the designer.

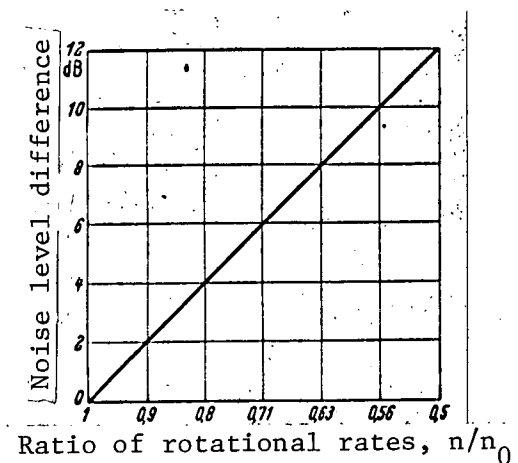


Figure 2. Noise level difference with change in the rate of rotation in planers.

Decreasing the rotational rate of the tool spindle and the working velocity of the delivery air in the suction systems produces a further diminution of noise. To be sure, the noise level does not drop off continuously with decreasing rotating rate in the machines because of the resonance spaces which exist, but there is a general dropping tendency. For machines with cutting spindles, the sound intensity, L , rises or falls with the fourth power of the spindle rotational rate:

$$L = 10 \cdot \lg \left(\frac{n}{n_0} \right)^4 \text{ [dB]} \quad (1)$$

With a 33% change in the rate of rotation, one obtains, with an increase in the rate of rotation, $L = 40 \lg 1.5 = 7 \text{ (dB)}$; and with a decrease in the rate, a decrease of $L = 40 \lg 1/1.5 = -7 \text{ (dB)}$ (Figure 2).

For electric motors and blowers, we have the rule of thumb

$$L = 10 \cdot \lg \left(\frac{n}{n_0} \right)^5 \text{ [dB]} \quad (2)$$

so that the sound level will drop by 15 (dB) if the rate of rotation is halved.

The flow noises in suction systems arise primarily through production of turbulence due to unfavorable outside shape. Also, the flow rate exerts a decisive influence. The sound intensity climbs with the fifth to sixth power of the flow rate. This means that if the flow rate is halved the noise is diminished by 15 to 18 dB.

In the combination of several working units into one machine, it is of importance that the increase in the total noise level from several equally loud noise sources, in comparison to the sound level of one such noise source, is

$$L_{\text{ges}} = 10 \cdot \lg \frac{I}{I_0} + 10 \cdot \lg n = L_0 + 10 \cdot \lg n \text{ [dB]} \quad (3)$$

The summation of 10 equally loud noise sources gives an increase in the total noise level by 10 (dB). It is perceived as twice as loud as the noise level of a single noise source (Figure 3).

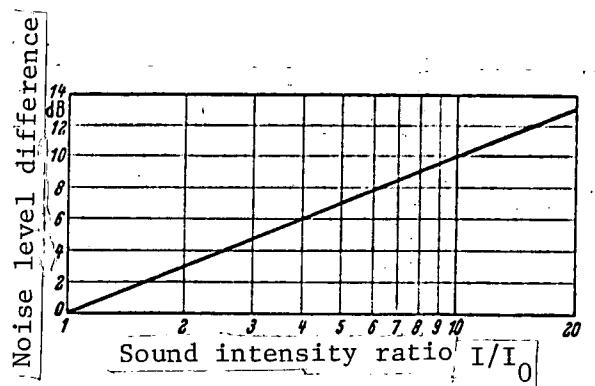


Figure 3. Noise level difference from summation of equally loud noise sources.

Figures 4 and 5 show the results which can be aimed at by decreasing the spindle rotational rate and by noise-blocking measures on a thickness planer for processing chipboard. The entire machine was installed in a chipboard box covered internally with a felt layer 70 mm thick. Its operating data were: work width, 1,250 mm; cutting depth, 2 mm; feed rate, 8 m/min; cutting rate, 30 m/sec (originally 46 m/s).

The oblique limiting lines in Figures 4 and 5 serve to judge the danger of hearing damage, which begins at limiting line 8. If the limiting line 8 is maintained, then the danger of hearing damage at the work area is excluded. In the present case, the total noise pressure level at the work place, only with sharp cutters, to be sure, was 90 (dB) (B) (= DIN phon). This corresponds to the value allowed by VDI 2058. The success of the measures was in the fact that the high-frequency vibrations, which are perceived as particularly unpleasant, could be suppressed.

But still other information and noise reduction measures are important for machines with cutting spindles. The siren tone in the operation of planing machines is produced by a pressure field rotating with the knife clamps and by pressure disturbances produced by the cutter along with the clamping bars, the table lips, and the work piece.

According to the studies of Pahlitzsch and Liegmann (1956), a decrease of the idling noise strength by 10 to 15 DIN phon is attained by pressure compensation by means of openings in the table lips. Pressure compensation in the axial direction of the cutting spindle and a decrease in loudness by 10 to 15 phons is attained during cutting with cutters set in a spiral. To be sure, the cutter spindle with twisted cutters could not displace the one with axially parallel cutters because of the difficulty of inserting and adjusting the cutters. Changes in the profile shape of the cutter spindle could not affect noise production.

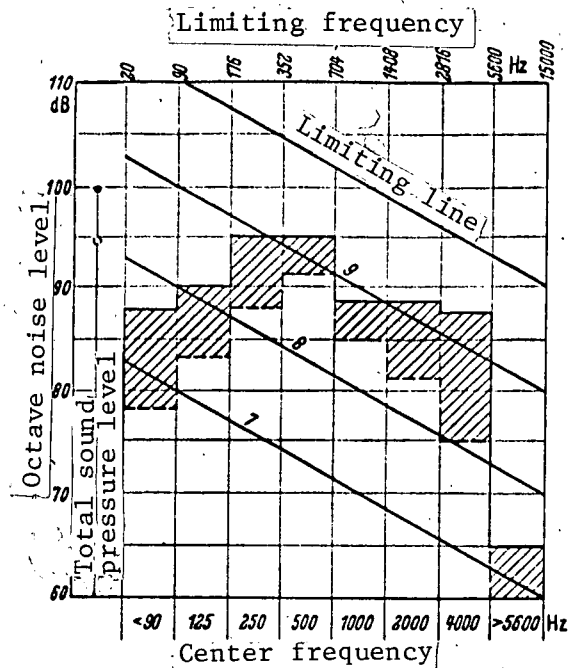


Figure 4. Work area noise level for a thickness planer. Dashed lines are for sharp cutters, and solid lines for dull ones.

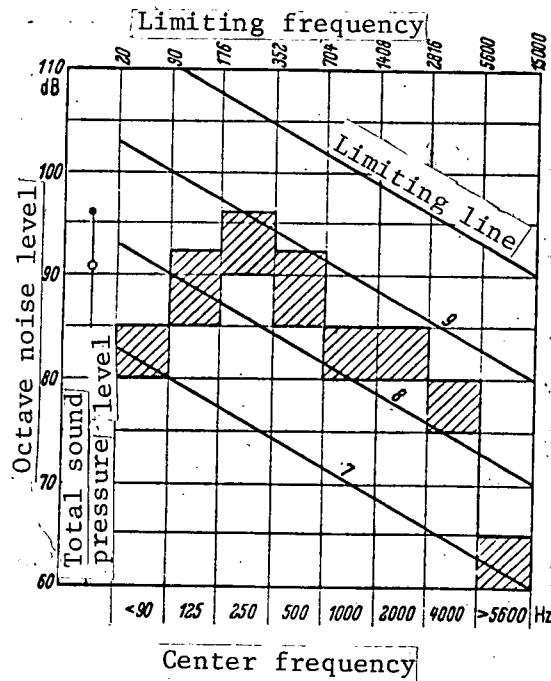


Figure 5. Work area noise level for a thickness planer after installing the machine in a box of sound-absorbing material.

Pressure disturbances during idling and during milling could be decreased with a balanced cutting spindle design (Schmutzler, 1965). With such a cutter shaft, slots are placed in the knife holders and in the carrier body at close separation. They form pressure compensation channels from the chip slots to the rest of the periphery. To be sure, the pressure field rotating with the cutting spindle is not affected with this design, so that it is not possible to reduce the loudness down to the running noise of a smooth shaft, which can be considered the lower limit.

Particularly severe noise occurs in thickness milling of large-area plates. It increases as the knives get dull. The powerful knife blows perpendicular to the plane of the sheet produce intense vibration in the sheet. The size of the vibrating plate favors sound radiation; i. e., conversion of body sound into air sound. The vibrations in the sheet can be weakened by installation of rubber cushions in the feed rolls and under the table plate. The frictional forces within the rubber cushions decrease the vibration intensity, especially for the high-frequency vibrations. The noise becomes duller, and is, therefore, perceived as less unpleasant.

CIRCULAR SAWS

In these machines, axial movements of the saw blades have proved to be particularly bad for noise production. These axial motions are produced by deviations in the shape of the saw blade, inherent vibrations of the saw blades, and by the teeth biting in. The shape deviations arise as a result of the stressed state of the saw blade and the static flat stress from the clamping flange faces. During operation a decrease in the flat stress on the blade occurs because of the stiffening effect of centrifugal force (straight line setting). The axial movement of the saw shaft is determined by the rigidity of the machine, the saw shaft bearings, and the drive. The inherent vibrations depend on the geometric dimensions of the saw blade, the type of clamping, and the internal stress condition. Additional vibrations can be produced during working by cutting force components perpendicular to the plane of the blade.

In idling, the high-frequency vibrations of the saw blade (inherent vibrations) which are heard as whistles become particularly unpleasant. But these whistles do not occur with all saw blades. Pahlitzsch and Rowlinski (1966) concluded from this that it is a matter of certain stress distributions resulting from dressing and prestressing, and that it is possible to prevent the whistling by producing a certain stress condition. Corresponding experiments showed that this is in fact possible.

Noise-damping measures for circular saws, which have for a long time been too little used, were recommended by Pahlitzsch (1961, 1962). A decrease in idling noise was attained by clamping the saw blade in a rotating damping disk having a diameter about $2/3$ that of the saw blade. The noise damping is apparently caused by a thin layer of air between the saw blade and the damping disk. The results are not yet satisfactory, as they do not produce a decrease in noise during the sawing itself, and the application remains limited to cut depths less than $1/6$ of the saw blade diameter.

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In idling, fixed damping plates placed below the circular saw table at the smallest possible distance at both sides of the saw blade (i. e., with about 2 mm air gap) decrease the noise. This is apparently also based on damping of vibrations in the rotating saw blade by the thin air layer between the saw blade and the damping plates. No noise-reducing action occurs on sawing. This might be due to saw chips striking the damping plates.

When plates are being trimmed, the trimmings which fall off are usually chopped up and sucked away by cutting tools placed immediately beside the circular saw blade, called trimming choppers. The frame of the trimming chopper immediately beside the circular saw blade damp the saw blade vibrations and decrease the idling noise. The effect of the trimming chopper on the air pressure is insignificant, so that an over-all decrease in noise can be registered during idling, in comparison to the circular saw blade running alone. During operation, to be sure, this advantage can easily be converted to a disadvantage by improper chopper design. The trimmings should be chopped in small lots by step-segmented

choppers. If the chopper blade strikes simultaneously across the entire chopper width, then, because the projecting trimming strikes the plate clamps, which tend to vibrate, there results a higher noise level than with the circular saw blade working alone. The trimming chopper is best equipped with a light metal carrier body and with hard metal coated cutting teeth inserted in steps at a raking angle.

Trimming choppers of flat, four-cornered steel plates with hard metal cutting teeth, arranged in a packet with the cutters staggered around the periphery, should also have a stepped cutter circle diameter, increasing from the saw blade outward, so that a noise decrease by 5 (dB) (B) can be attained in comparison to the unstepped one.

SUCTION REMOVAL OF CHIPS

Suction removal can become an irritating noise source with improper shape of the chip hood, excessive pressure losses in the machine, blocked induction of carrying air, too-long metal hoses at flexible connections, unfavorable flow designs of pipe bends and branches, and excessive working velocity.

The following basic principles should be noted in the design of chip suction systems.

Chip catchers and protective hoods should be designed so that the connecting tube is in the flight direction of the chips. At least, the main direction of flow should be in the flight sector for the chips. In this way, the kinetic energy of the chips can be used to carry them farther.

The cross-sectional area of the connection should be designed with the same area as the open suction cross section. There should be no constrictions or expansions of the cross section in the collecting hood. Overly small connecting cross section and cross section constrictions are particularly bad for noise, as they produce pressure losses and, due to excessive flow velocity, aerodynamic noise.

If coarse material falls in along with chips, there should be a coarse material separator in the machine or suction system. Carrying of coarse material may require working velocities over 25 m/sec in the pipelines, causing severe working noises.

With high suction capacity, carrier air openings should be provided at the machine, to draw in outside air through pipes or base channels. Attention should be given to a favorable shape for flow technology.

Metal hoses should be kept as short as possible because of the greater pressure losses in comparison to smooth pipes.

Chip blowers can also become noise generators, especially if the blower wheel is not dynamically balanced, or if it has become unbalanced by bent or missing blades.

In the design of blowers, it must be noted that mechanical noise is radiated in the air and transfers mechanical vibrations to the pipe lines and the foundation. By installing the blower with vibration isolation, and by making the suction, pressure, and collecting hood connections through a 10 to 20 cm section of rubberized cloth hose to isolate noise conduction through the solid, vibrations and mechanical noise can be decreased.

If a high flow velocity is necessary in the supply pipes, the noise transferred through the air line must be weakened by insulation of the air channel with sound-absorbing materials or by using sound dampers.

CUTTING SPINDLE MILLS

At the control position of cutting spindle mills with completely welded construction and cutting spindles with axially parallel cutters, the noise evaluation number of N 100 according to TGL 10687 will be exceeded. These machines are, along with automated double end profile mills, the most unpleasant noise sources in the wood industry.

The noise originates through the cutting spindle, from impacts of the knives on the material being milled, especially with cutters which are nearly dull, and by conduction of sound through the solid by welded components, base plate, and stand. Sympathetic vibration of large-area machine stands can be avoided only with difficulty. Maintenance and tool changing are hindered by a noise-absorbing covering. The total noise level can be diminished by use of gray cast iron for the major components and by cutting with a rake angle. The unpleasant high-frequency vibrations are suppressed.

EVALUATION OF LOW-NOISE INTERVALS

In relation to the noise reduction attained in idling for various machines, the allowable period of action and the intervals between two noise actions are important. The allowable limits for noise action are summarized in TGL 10687. According to this standard, the noise evaluation number N 85 must not be exceeded when there is uninterrupted action of a broad-band noise during a working day of 5 or more hours (Figure 6). When there is

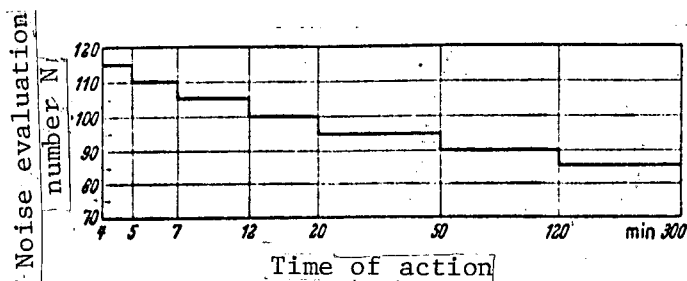


Figure 6. Acceptable noise action periods at various noise evaluation numbers according to TGL 10657.

intermittent action, Figure 7 shows the relation between periods of action, noise intervals, action times, and noise evaluation numbers. The limits are given by the intersection points for the 4 influencing quantities.

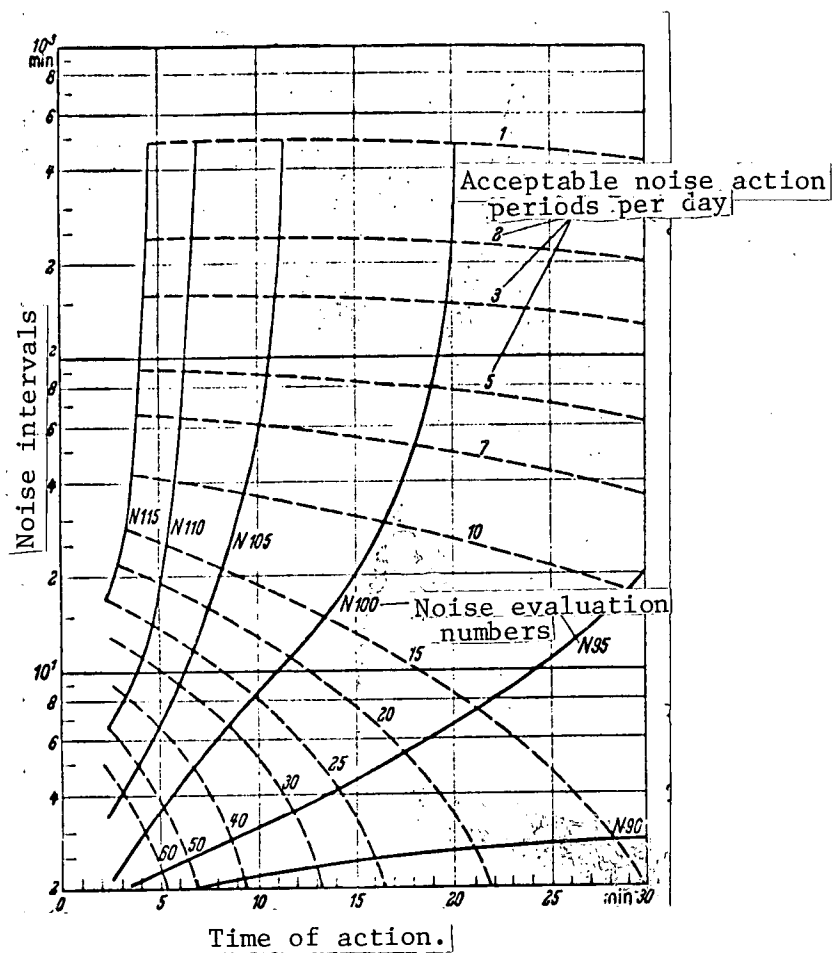


Figure 7. Relations between the times of action, noise intervals, number of noise action periods, and noise evaluation numbers according to TGL 10687.

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